THE INSERTION LOSS OF POROELASTIC PLATE SILENCERS IN A FLOW DUCT

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1 INTRODUCTION

Passive noise control in ducts has been achieved using structural, reactive and dissipative devices [1 - 7]. Huang [1,2] has carried out theoretical and experimental work on structural devices involving the duct walls. Ramamoorthy *et al* [3] have used experimental and theoretical methods to investigate the insertion loss of a passive structural acoustic silencer taking the effect of the external fluid into account. Also they have presented an interesting relationship between the insertion loss and plate dispersion. Tang and Lin [5] have studied the resonant mass-spring behaviour of a stiff light composite panel absorber, and have investigated its applicability for noise reduction. Astley *et al* [6] have examined some of the effects of wall flexibility on sound propagation and attenuation in a duct with a bulk porous liner and negligible mean flow. Cummings and Chang [7] have presented models that describe the effect of internal mean flow on the bulk acoustic properties of a porous medium and have predicted the effect of the absorbent *on* the sound transmission loss of the silencer. Typically dissipative silencers are designed to exploit the most widely- acknowledged absorption mechanism in porous materials viz. viscous friction due to relative motion between solid and fluid. If the frame of the porous material is viscoelastic then other dissipative mechanisms are possible. To obtain good low frequency absorption, say below 300 Hz, exploiting the viscous friction mechanism alone may require an unacceptably large thickness of material. On the other hand, when designing passive sound absorbers based on porous materials or membranes, it is known that the presence of a backing air gap with properly chosen dimensions can enhance their low frequency performance. Furthermore, if an elastic porous plate is clamped at its edges, then a backing air cavity will allow bending modes which could lead to significant absorption through viscoelastic losses at relatively low frequencies. Indeed useful low frequency absorption maxima have been observed for configurations consisting of clamped poroelastic plates with an air gap behind them [8]. It would seem likely that the triple combination of bending vibration, the presence of an air gap and the higher frequency visco-thermal sound attenuation in a duct-wall silencer using a porous elastic plate with an airgap might give useful acoustic performance over a wide frequency range. Perforated plates are used in exhaust silencers, noise attenuators in ducts, and duct linings in jet engines. Recently Aygun *et al* [9,10] have investigated the effects of inserting a perforated porous plate placed transversely across the duct on the uniformity of flow and the sound absorption in the duct. These effects were assessed by measuring the insertion loss at different locations in the duct. A parallel impedance model was used to model the effects of perforation.

This paper presents measurements of the sound insertion loss (TL) in a flow duct due to simple systems consisting of clamped non-perforated and perforated porous elastic plates with an air cavity backing. The effects on the acoustic performance of the porous elastic plate silencers of perforations and flow rates are investigated. The open cell foam plates were made from recycled materials at the University of Bradford. Each of the porous elastic plates were mounted on fairings near to the duct wall parallel to the direction of the air flow and separated from the walls by an air cavity. The purpose of separating the plate from the walls by an air cavity is to allow bending vibration which should result in attenuation due to viscoelasticity as well as flow resistivity. Separating the plate from the wall by an air cavity causes the resonance frequencies of the plate to increase because the stiffness of the system is increased by the air cavity. In mean flow, the poroelastic plate silencer is excited by lateral components of the air flow and by acoustic plane waves produced from loudspeakers. A fraction of the acoustic energy vibrates the plate in flexural modes. The resulting vibrational energy is partly converted into heat and dissipated in the structure of the porous frame and partly re-radiated at the resonance frequencies of the plate. Another fraction of the sound energy is dissipated through the visco-thermal interactions between the solid and the fluid in the pores. Vibration losses have the greatest effect at low frequencies while the visco-thermal dissipation has a significant effect at high frequencies.

The sound insertion loss has been calculated from the differences between averages of several sound pressure measurements either side of the porous plate structure in the flow duct. To investigate the contributions of plate vibrations to the measured IL, the displacements of a perforated and non-perforated porous elastic plates separated from duct wall by an air cavity in the duct have been measured and compared with those of the duct wall.

2 MEASUREMENT SYSTEM

2.1 Construction of the clamped plate silencer

A plan view of the porous plate silencer is shown in Figure 1. A 20 mm thick poroelastic plate with dimensions of 1 m x 0.8 m is mounted parallel to one side of the flow duct, and separated from the wall by an air cavity. The length and breadth of the cavity are the same as those of the porous plate, and the cavity depth is 60 mm. The plate is mounted parallel to the direction of the air flow.

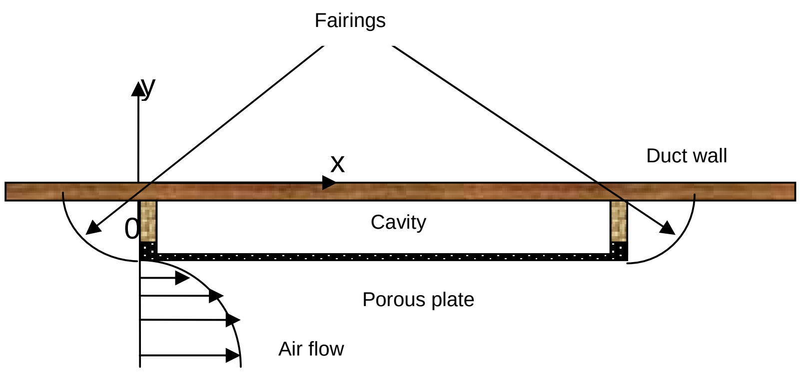

Figure 1 Plan view of porous plate silencer mounted on duct wall

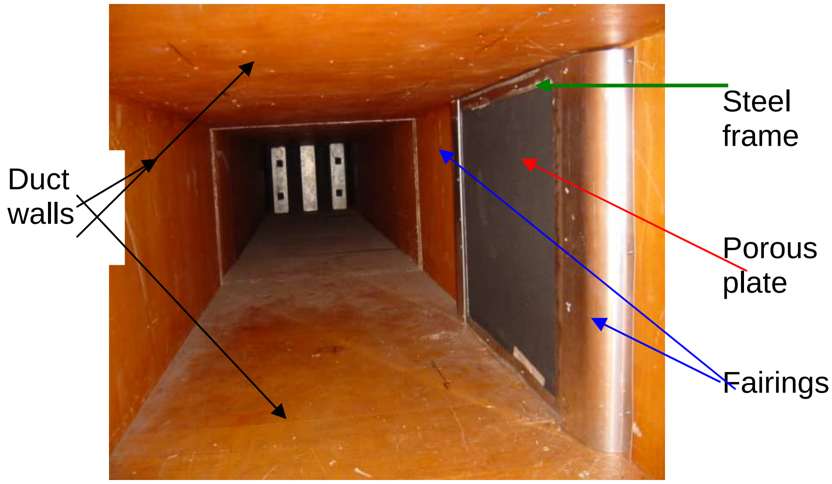

Figure 2 A 1m × 0.8m × 20mm thick porous elastic plate mounted in fairings on a side wall in the Duct.

A steel frame having 0.7 m x 0.9 m internal dimension with 3 mm thickness is used to hold the plate and reduces the dimensions of the exposed area of the plate to 0.9 m x 0.7 m. Two steel fairings are attached to edges of the plate to encourage the sound waves and air flow to travel in the direction parallel to the plate. There are two fluid/plate interfaces at the front and rear of the plate and the plate is excited by the incoming acoustic plane wave and the air flow. Figure 2 is a photograph of the porous plate mounted in the steel fairings on the wall of the flow duct.

2.2 Flow Duct Measurement System

The flow duct measurement system is shown in Figure 3. The internal cross-section of the duct is 1.2 m x 0.8 m. By means of the fan impeller, volume flow rates of up to 17 m^3/sec can be achieved at a pressure differential of approximately 120mm water across the fan inlet. The flow rate is changed by varying the fan blade pitch angle between -2^0 and $+24^0$. As well as the noise caused by the fan and air flow, sound is created inside the duct by four 600 W bass loudspeakers fed with random noise filtered in octave frequency bands from a B&K noise generator, type 1405, and a 1.6 kW power amplifier, type SR707. A ½ in B&K microphone, type 4133, fitted in a B&K turbulence cancelling tube, type UA0436, is used to sample the acoustic sound field. The microphone is supported by a preamplifier that is connected to a B&K measuring amplifier, type 2609. The output of the measuring amplifier is fed to a two channel FFT analyzer, type CF350Z. The signal to noise ratio was between 12 dB and 17 dB for measurements made in the presence of the silencer, and was between 12.5 dB and 16.5 dB for measurements made without the silencer.

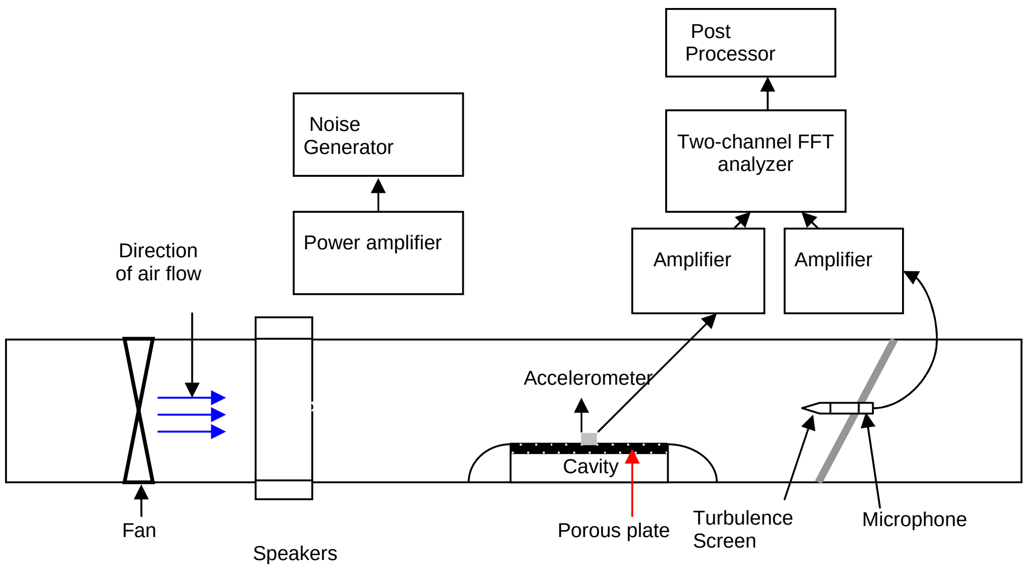

Figure 3 Flow duct measurement arrangement; dimensions in the figure are not to scale.

The porous elastic plate, made of recycled car dashboard materials, is clamped between a steel frame and a wooden support. The properties of this plate have been measured at the University of Bradford and are given in Table 1. It should be noted that the insertion loss results reported here are obtained with the clamped plate mounted on one side of the duct as shown in Figure 2. When the clamped plate system is mounted at the top or bottom of the duct, the plate bends as a result of self-loading and this affects the insertion loss. The IL observed when the plate was mounted at the top or bottom of the flow duct was negative throughout the frequency range.

The perforations consisted of 9 × 42 mm diameter holes giving an area perforation ratio of 1.6%.

Table 1: The assumed characteristics of the porous plate

L_x (m)	L_y (m)	Thickness h (mm)	ρ (kg/m^3)	E(Pa)	Loss factor	Porosity ϕ	Poission ratio ν	Flow resistivity, Ns/m^4	Tortuosity α_∞
0.9	0.7	21	223	2.46x10^6	0.35	0.75	0.3	46933	1.85

3 RESULTS

The procedure described elsewhere [9, 10] and the recommendations given in ISO standard 7235 [11] have been followed during measurements. The insertion loss (IL) of the poroelastic plate silencer has been calculated from sound pressure measurements in the duct according to [11]:

$$IL = \overline{L}_{pII} - \overline{L}_{pI} \tag{1}$$

where $\overline{L}_{pI}$ is the spatial average sound pressure level in the frequency band in the test duct, when the test silencer is installed, and $\overline{L}_{pII}$ is the spatial average sound pressure level in the frequency band in the test duct, when the silencer is not present.

The sound pressure level, $\overline{L}_p$, has been calculated by measuring the local sound pressure levels at least at three key positions equally spaced on the longitudinal span of the line. The spatial average sound pressure level, $\overline{L}_p$, in dB, has been determined from the local sound pressure levels, L_{pi}, using [11] :

$$\overline{L}_p = 10\log\left[\frac{1}{n_m}\sum_{i=1}^{n_m}10^{\frac{L_{Pi}}{10}}\right] \tag{2}$$

where n_m is the number of measurements.

The measured IL spectra of non-perforated and perforated poroelastic plate silencers without mean flow but with noise from the loudspeakers are shown in Figure 4. The maximum IL when the plate is not perforated has a value of 4.1 dB at 1 kHz. With perforations, the maximum IL is 3.2 dB at 630 Hz. The measured IL is below zero at both low and high frequencies. At high frequencies this may be the result of the fact that the waves might not be plane and there may be flanking paths. At low frequencies the apparently negative IL might be the result of axial reflections from the fairings or damped axial resonances of the porous plate. Only positive insertion loss values are shown in these and the subsequent graphs.

IL measurements have been carried out also with non-perforated plates in the presence of average air flow speeds of 5.5 m/s and 37.65m/s in the duct, corresponding to Mach number (*M*) values of 0.016 and 0.11 respectively but without the loudspeakers operating so that the noise in the duct was due only to the air flow, the fan and background noise. The resulting IL spectra are shown in Figure 5. The measured IL with a mean air flow speed corresponding to *M* = 0.11 has a peak of 16.4 dB at 250 Hz. Above 500 Hz, the measured IL values for the higher Mach number flow are below zero. Increasing the air flow results in an increase in the maximum IL and a shift of the maximum IL to a lower frequency.

IL measurements for perforated and non-perforated porous plate silencers have been carried out in presence of both mean air flow (*M* = 0.056 and 0.11) and noise from the loudspeakers. Resulting data are compared in Figures 6(a) and (b). Figure 6(a), for the lower flow rate shows a mostly above zero IL with a peak of 5.4 dB for the non-perforated porous elastic plate at 3150 Hz. The IL peaks observed at frequencies below 1000 Hz with perforation are higher than IL peak observed without perforation. At the higher flow rate, Figure 6(b) shows that the IL for perforated and non perforated porous elastic plates are similar above 100 Hz. However, the peak IL at 160 Hz is significantly higher (11.5 dB without perforations and 11 dB with perforations) than any observed at the lower flow rate. At frequencies below 100 Hz for the higher flow rate, the IL measured for perforated plate has been found to be higher than that obtained using the non-perforated plate.

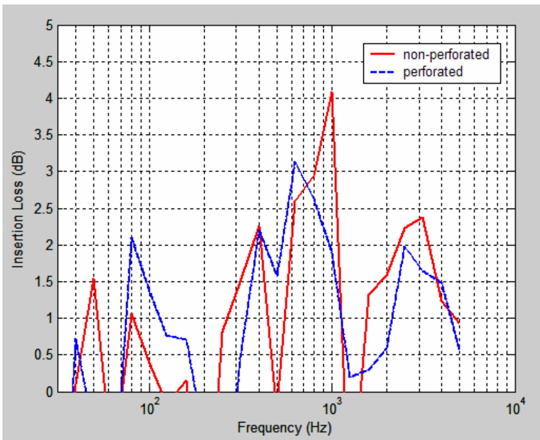

Figure 4 Measured insertion loss of a perforated and non- perforated porous plate without mean flow but with noise

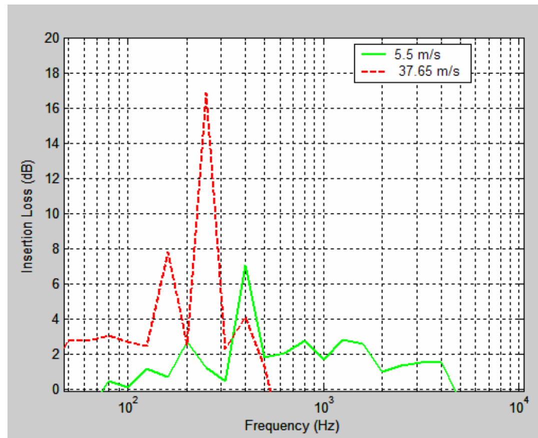

Figure 5 Measured insertion losses of non-perforated poroelastic plate silencers with mean air flow at speeds of 5.5 m/s and 37.65 m/s.

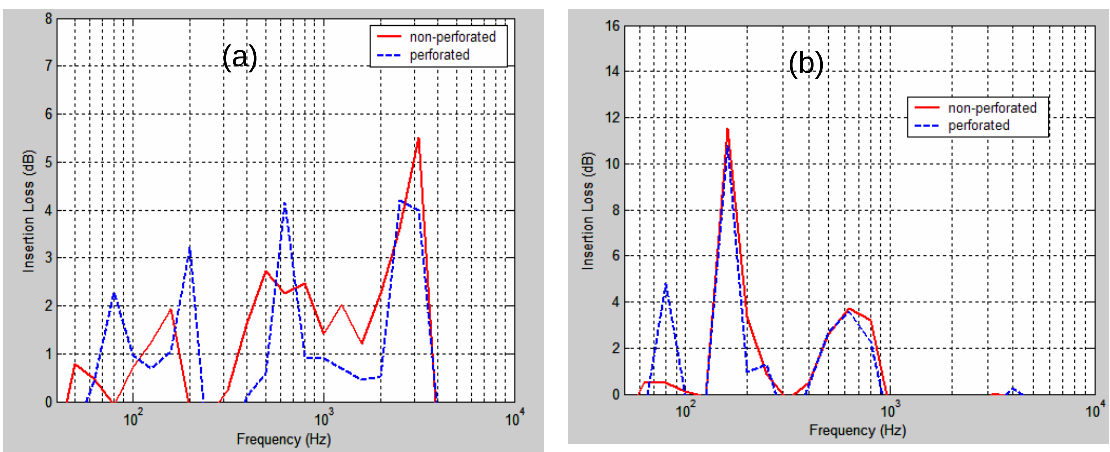

Figure 6 Measured insertion loss of perforated and non-perforated porous plate silencers in the presence of noise from the loudspeakers (a) with the lower air flow rate (M = 0.056) and (b) with the higher rate of air flow (M = 0.11).

Overall (see Table 2), it seems that increasing air flow velocity will decrease the insertion loss of poroelastic plate silencers at high frequency and increase their insertion loss at low frequency. Somewhat counter-intuitively, perforating the plate results in higher insertion loss peaks at low frequencies (see Figures 4 and 6 and Table 2).

To investigate the extent to which the poroelastic plate silencer performance depends on the plate deflection, measurements have been performed of the deflection of the perforated and non perforated poroelastic plates in the silencer arrangement. A B&K accelerometer, type 4374, was used to monitor the acceleration of the plate. The accelerometer was fixed to the plate by means of double side tape. The output of the accelerometer is connected to a B&K measuring amplifier, type 2609 (see Figure 3).

Table 2 Magnitudes and frequencies of peaks in Insertion Loss for Perforated Plate (PP) and Non-Perforated Plate (NPP) systems with and without noise and air flow corresponding to Mach number (M) values of 0.056 and 0.11.

Perforation, Noise and Flow Conditions	**Frequency (Hz)**											
	40	**50**	**80**	**160**	**200**	**250**	**400**	**500**	**630**	**1000**	**2500**	**3150**
Peaks in IL with noise (dB), Non Perforated Plate (NPP)		1.5	1				2.25			4		2.4
Peaks in IL with noise (dB), Perforated Plate (PP)	0.75		2				2.2		3.2		2	
Peaks in IL with air flow (dB), NPP, M = 0.016					2.8		7					
Peaks in IL peaks with air flow (dB), NPP, M = 0.11				8		17	4					
Peaks in IL with air flow and noise (dB), NPP, M = 0.056		0.9		1.9				2.8				5.5
Peaks in IL peaks with air flow and noise (dB), PP, M = 0.056			2.2		3.2				4.1		4.2	
Peaks in IL with air flow and noise (dB), NPP, M = 0.11				11.5					3.6			
Peaks in IL with air flow and noise (dB), PP, M = 0.11			4.8	11					3.6			

The displacements measured at the centre of the porous elastic plate are plotted versus frequency in Figures 7(a) and 7(b) for three different cases. The measured displacements of the non perforated poroelastic plate, with and without noise, in the presence of air flow at a speed of 5.5 m/s are similar to each other. This means that the insertion of noise from the loudspeakers does not change the displacement of the plate very much when there is air flow. The measured displacements of the perforated poroelastic plate with and without noise, in the presence of air flow, are similar to each other (see Figure 7(b)). However perforation decreases the displacement of the poroelastic plate.

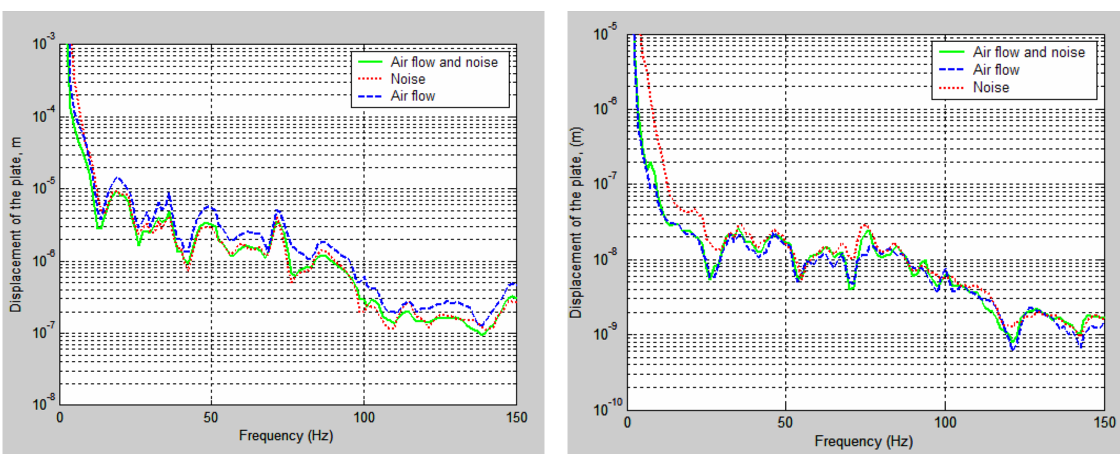

Figure 7 Measured insertion loss of perforated and non-perforated porous plate silencers in the presence of noise from the loudspeakers (a) with the lower air flow rate (M = 0.056) and (b) with the higher rate of air flow (M = 0.11).

To investigate the extent to which there may have been structure-borne components of the plate deflection, the displacement of the duct wall has been measured and is plotted versus frequency in Figure 8. The displacement of the duct wall is comparable with the displacement of the perforated porous plate. The displacement curves are mostly similar except for the resonant peaks observed at 35 Hz and 75 Hz which are close to the frequencies at which the insertion loss of the porous plate is found to be below zero. The measured peak deflection frequencies (in Hz) for the porous plate and duct wall are shown in Table 3.

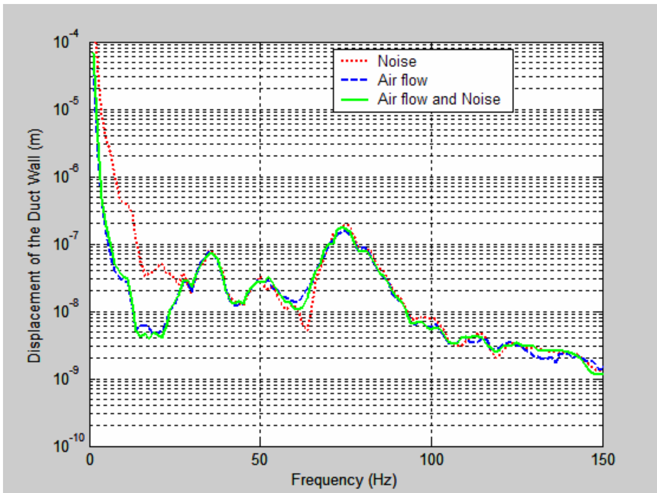

Figure 8 Measured displacement of the Duct wall.

Table 3: Experimentally-observed resonance frequencies (Hz) for the deflection of the porous plates mounted with an air cavity on the duct wall.

Non-perforated plate										
Air flow and noise	18.75	36.25	48.75	61.25	71.25	86	102.5	113.5	130	148.8
Air flow	18.75	36.25	48.75	62.25	71.25	86.25	100	115	127.5	148.8
Noise	18.75	36.25	50	60	71.25	86.25	101.3	115	126.3	
Perforated plate										
Air flow and noise	7.5	35	47.5	62.5	67.5	76.25	85	93.75	100	128.75
Air flow	7.5	36.25	46.25	62.5	67.5	76.25	86.25	100	127.5	
Noise	21.5	36.25	46.25	62.5	67.5	75	83.75	115	128.75	147.5
Duct wall										
Air flow and noise	11.25	18.75	28.75	35	48.75	52.25	73.75	112.5		
Air flow	11.25	16.25	27.5	35	48.75	52.5	73.75	80	96.25	101.25
Noise	11.25	21.25	27.5	35	50	53.75	60	75	81.25	113.75

4 CONCLUSIONS

The insertion loss spectra of perforated and non-perforated poroelastic plate silencers mounted parallel to a duct wall without and with air flow have been deduced from sound pressure measurements versus frequency. The results show that, in the presence of airflow, perforation increases the low frequency IL. It has been observed that, whether or not the plate is perforated, increasing air flow velocity decreases the insertion loss at high frequency and increases the insertion loss at low frequency. The plate and duct wall deflections have been measured also. The introduction of air flow in the flow duct has been observed to increase the plate deflection. In the case of the non-perforated plate this is consistent with the hypothesis that the plate deflection is due to the lateral components of the air flow and the acoustic plane wave. In the presence of air flow, perforation has been found to increase the insertion loss of the plate at low frequencies but to decrease the displacement of the plate in comparison with non perforated poroelastic plate. Measurements of the displacement of the duct wall show that, in the case of the perforated plate, the plate deflection is mainly structure borne which suggests that the low frequency improvement is associated with the perforations.

5 ACKNOWLEDGEMENTS

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